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SOME PRELIMINARY RESULTS OF BRUSH SEAL/ROTOR INTERFERENCE EFFECTS

ON LEAKAGE AT ZERO AND LOW RPM USING A TAPERED-PLUG ROTOR

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Abstract

Some preliminary brush seal leakage results for ambient-temperature air are presented. Data for four nominal brush-rotor radial clearances of -0.09, -0.048, -0.008, and 0.035 mm (-0.0035, -0.0019, -0.0003, and 0.0014 in.) were taken by using a tapered plug rotor at 0 and 400 rpm with rotor runout of 0.127 mm (5 mils) peak to peak. The brush seal nominal bore diameter was 38 mm (1.5 in.) with 0.05-mm (2-mil) bristles at 200 bristles/mm of circumference (5000 bristles/in. of circumference) and a 0.61-mm (24-mil) fence height. Leakages were greater than predicted, but agreement was reasonable. Leakage rates were not significantly altered by hysteresis or inlet flow variations. Visualization studies showed that the bristles followed the 400-rpm excitation, and loading studies indicated that bristles slid relative to one another.

Introduction

In many aircraft gas turbine engines labyrinth seals are being replaced by brush seal systems because brush seals are compliant and reliable, leak less, cost less, and enhance rotor stability. Brush seal systems consist of the brush and a hardened rub ring. Typically, the bristles are oriented to make an angle of 30° to 50° with the rotor radius. This design allows the bristles to flex when rotor excursions occur and decreases bristle tip loadings. Because brush seals are contact seals with radial interferences ranging from zero to more than 0.25 mm (10 mils), ceramic coatings are often used on the rub ring, and the bristles are made of superalloy materials to enhance life and minimize wear at elevated surface speeds, temperatures, and pressure drops.

The brush seal configuration can be linear, circular, or contoured. Circular brush seals manufactured by Cross Mfg. Ltd. (Fig. 1) and Rolls-Royce have been successfully operated in jet engines for hundreds of hours.[1] The leakage through a brush seal is 1/3 to 1/2 that of a four- to five-cavity labyrinth seal.[2,3] Others claim the leakage to be 1/10 to 1/20 that of the labyrinth seal, but without citing the geometric configurations of either the labyrinth or the brush seals. In addition, a brush seal costs less to manufacture. The projected production costs of a circular brush seal approach $40/cm of diameter ($100/in. of diameter) plus tooling, which is perhaps 1/5 that of carbon or labyrinth seals. The seal system costs when accomplished by a competent manufacturer approach $60/cm of diameter ($150/in. of diameter) for a 10- to 15-rms surface finish of Al_2O_3, CrO, or magnesium-zirconate on a high-Ni-based alloy substrate. Finally, although seal dynamic perturbations are a major concern, Conner and Childs[4] found a four-brush seal configuration to be more stable than a labyrinth seal.

Although brush seals show great promise for future applications, it must be acknowledged that they are contact seals in which life and wear rates are a major concern. They are also limited by pressure drop, which if too large will cause the bristles to blow out.[5,6] Difficulties have been experienced in applying brush seals to contours other than simple circles, particularly those with convex curvatures. Several manufacturers (Cross Mfg. Ltd., Technetics, and Sealol) are known to fabricate both concave and convex brush seals. In addition, Textron-Lycoming has three-dimensional woven brush configurations, and Steinetz et al.[7,8] and the 1991 Seals Workshop[9] have reported on a three-dimensional, woven-rope ceramic seal as well as a segmented-platelet ceramic seal. Currently, brush seals are limited to ambient and hot gas applications. Proposals for the use of brush seals in cryogenic turbomachines are being investigated.[10]

Analytical modeling of brush seals has been limited. The simplest model uses an aperture with the proper flow coefficient. Bulk flow models[11,12] and a preliminary numerical model[13] have also been developed. Some experimental results are also available.[2-6] Visualization studies have been carried out for simulated linear brush configurations in water and oil tunnels.[11,14-16] The major observed flow patterns were lateral (normal to the bulk flow) and highly interrupted by the bristles. Cross Mfg. has provided some leakage results for radial clearances varying from 5 to -5 mils for a 5.1-in.-bore-diameter brush seal with a 70-mil fence height (backing washer clearance) and 2.8-mil bristles at 2500/in. of circumference. These results and a comparison with theory are repeated herein. Although the authors are aware that other leakage data exist for ranges of interference fits and other configurations, these results are proprietary.

This paper presents some preliminary leakage data for air flowing through a 37.9-mm (1.4926-in.) bore-diameter brush seal at four radial clearances: -0.09, -0.048, -0.008, and 0.035 mm (-0.0035,

-0.0019, -0.0003, and 0.0014 in.). A tapered-plug rotor installed in a drill press was used to obtain these clearances. Data were taken at 0 and 400 rpm over a range of pressure drops from 0 to 0.83 MPa (120 psid). Maximum and minimum pressure drops associated with rotor runout, which was less than 0.13 mm (5 mils) peak to peak, were also measured at 0 rpm. Owing to runout, continuous bristle contact with the rotor was assured only for the -0.09-mm (-0.0035-in.) radial clearance data. Some hysteresis was noted and the data were compared with theory.

Apparatus

The apparatus consisted of a small pressure vessel that both held the brush seal in position and provided a circumferential seal at the outside diameter, a retaining flange that served to hold the brush seal in position, and a plug-shaped rotor with a machined drill bit drive (Fig. 2). The apparatus was mounted on a base plate that was clamped to a drill press table to provide a stable platform; vertical motion was controlled by the drill press arbor. The seal clearance or interference was changed by adjusting the vertical position of the tapered plug. Service air was supplied to the brush seal through a single inlet near the base of the pressure vessel. The inlet flow rate was measured through one of three parallel venturi flowmeters. The inlet pressure and temperature were measured near the inlet to the pressure vessel. Barometric pressure was also measured and added to the appropriate pressures.

Because the flow entered the pressure vessel at one port, concern over the uniformity of flow at the inlet prompted the addition of a flow straightener consisting of a bundle of 6-mm (0.25-in.) long soda straws.

The nominal 38-mm (1.5-in.) diameter brush seal was fabricated by Cross Mfg. The seal outside diameter was 53.33 mm (2.0996 in.); the nominal inside diameter, 37.9 mm (1.4926 in.); and the fence height, 0.61 mm (0.024 in.). The

0.051-mm (2-mil) long bristles were
Hastelloy-X, with 19 to 21 bristles in
the axial (or flow) direction and approx-
imately 9.84/mm (250/in.) in the circum-
ferential direction. A photograph,
closeup details, and a sketch of the
brush are given as Figs. 1 and 3.

The 304 stainless steel, tapered-
plug rotor had two tapers. The upper
tapered from 40.15 to 39.1 mm (1.5009 to
1.5394 in.) in diameter over 6.36 mm
(0.25 in.) and mated with the taper in
the pressure vessel's retaining flange to
provide the seal-rotor alignment func-
tion. The lower tapered from 38.6 to
37.8 mm (1.5202 to 1.4877 in.) diameter
over 25.36 mm (0.9986 in.). A photograph
and a sketch of the tapered plug are pro-
vided in Fig. 4.

The tapered plug was selected so
that information could be obtained at
various brush seal interferences and
clearances and so that the results could
be compared with cylindrical surfaces
(e.g., bore seal). It was also selected
as a useful way to change leakage with
rotor axial position.

At a later stage in the program a
halogen lamp was installed inside the
pressure vessel and a dual video camera
configuration was set up to view the
bristle-rotor interface (Fig. 5). The
video cameras, indexed to a portable
video switcher, simultaneously recorded
interface motion at two positions 180°
apart on a split screen. The images were
reflected in a mirror to a series of
extension tubes and lenses that magnified
the image 50 times before it was recorded
on 3/4-in. VHS NTSC video tape. The lamp
provided sufficient illumination of the
bristle-rotor interface so that both the
clearance and some bristle distortions
could be seen. With eccentric rotation
the illumination alternated between
intense broad bands of light, slivers of
light, and darkness. It was necessary to
sight along the bristle-rotor interface
to see the motion.

For all tests described herein the
working fluid was ambient-temperature
air.

Procedure

Prior to each run the transducers
were calibrated and the system was purged
with air. The oil-pump-compressed air
was filtered and a control valve was
opened to pressurize the seal. The pres-
sure was increased in increments and then
decreased to determine the effects of
hysteresis. Data were taken with and
without rotation.

Brush Seal Analysis

In previous papers the authors have
discussed flow-modeling parameters and a
method for predicting flow and pressure
drop.[11,12] This analysis is summarized in
Appendix A. The basis of the method is
flow through packed fibers or porous
media.

The model assumes a geometric
description of the brush seal. It also
assumes manufacturing tolerances and ele-
mentary bristle contact. Deformation of
the bristles is taken into account, but
interbristle friction is not. Some
effort was taken to account for packing.
Leakages at the interface and parallel to
the bristles and along the packing washer
interface were considered. The constants
are heuristic and matched to the results
obtained by Cross Mfg. Alterations for
the 38-mm (1.5-in.) diameter seal have
been necessary and are cited in the
Appendix A.

The analytical predictions of volu-
metric flow rate and pressure drop as
functions of clearance for both the 130-
and 38-mm (5.1- and 1.5-in.) diameter
brush seal configurations are given as
Fig. 6.

It is interesting to note that other
researchers have successfully correlated
data by using other techniques. Chupp[5]

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used the parameters flow rate and difference in the squared pressure $(P_0^2 - P_e^2)$ to present his results. The latter required the perfect-gas equation of state, which is valid for air at elevated temperatures and moderate pressures. The data predictions were empirical. Holle[6] presented his results in a similar manner and suggested that the experimental data can be predicted by using a flow coefficient approach. This approach proved to be an expedient and effective method of design. Steinetz et al.[7,8] used Kozeny-Carmen model parameters and measured weave resistances to make an equivalent flow resistance model that successfully predicted the flow in linear rope and platelet types of ceramic seals. The flow details of Lycoming's woven seals presumably can be assessed by a combination of the Hendricks et al. and Steinetz et al. work.[7,8,11,12]

Results

Leakage Experiments

Measured and predicted leakage as a function of pressure drop data for a 38-mm (1.5-in.) diameter brush seal operating at four clearances without inlet flow straighteners is given in Fig. 7. The data are for 0 and 400 rpm. Results are given for radial clearances of -0.09, -0.048, -0.008, and 0.035 mm (-0.0035, -0.0019, -0.0003, and 0.0014 in.).

As the clearance decreased, leakage decreased as expected, although the decrease tended to be larger than predicted, indicating that our model requires some modifications and a better understanding of the flow physics.

Some improvement in performance was seen at 400 rpm over 0 rpm (for an arbitrary position of the rotor), and the leakage data were in better agreement with the analysis (rotation tends to average out perturbations such as runout and seal tolerances).

For each clearance some hysteresis was noted, and it differed for the 0- and 400-rpm cases. It was also found that adding a flow straightener to the pressure vessel tended to reduce seal performance. But the opposite effect was noted for decreasing pressure with the flow straightener in place (Fig. 8). Neither of these effects on leakage were large and the dynamics could be altered, but Conner and Childs[4] found no significant changes in dynamics for large changes in inlet swirl.

Visualization Experiments

In a separate apparatus bristle behavior in a loading machine was visualized for a stationary rotor. Radial movement of the brush seal caused bristle packing that varied with angular position. This in turn caused the bristles to slide relative to one another with flexural displacement.

In the tapered-plug apparatus visualization of the bristle behavior at 400-rpm indicated that the bristles followed the interface motions. The extent of changes in interface clearance could not be quantified, but light variations in the circumferential direction qualitatively demonstrated nonuniformity in bristle spacing and contact at the interface. Distortions due to shaft or housing angular misalignment or casing motion were easily detected. Also noted were regions of excess bristle abuse where the bristles were not in contact at the interface. Such "holes" led to puffing sounds and jets of fluid through the seal. The videotape presentation is described in more detail in Appendix B.

The light visualization technique provides an excellent method for assessing the uniformity of rotor-stator clearances and determining interface distortions due to such operating parameters as temperature, pressure, rotation, and excitation. The lamps can be operated in

most environments provided that they come to equilibrium with that environment.

Summary

A tapered-rotor apparatus was constructed to obtain some preliminary brush seal leakage data with ambient-temperature air at four brush-rotor clearances for a 38-mm (1.5-in.) diameter brush seal. Although the leakages were greater than predicted, the agreement was reasonable.

Hysteresis was found for each configuration. Inlet flow straighteners also altered the leakages. Neither of these effects represent large changes in leakage, but they may affect the dynamics.

Visualization studies indicated that bristles packed but slid relative to one another with radial shaft motion and that at 400 rpm the bristles tended to follow the rotor motion.

The light visualization technique can be used to assess the uniformity of rotor-stator clearances and to determine interface distortions.

Appendix A

Abstract of Flow Model

The flow model centers around flow through fibrous materials and is described in more detail in references 11 and 12.

The primary leakage paths are at the interface, normal to the throughflow path, and parallel to the throughflow path. The latter two constitute flow through an anisotropic porous material, and the interface has various degrees of equivalent roughness.

Bulk Flow Model Hydraulic Diameter

$$D_v = 4 \text{ (Open volume/} \quad \text{Friction surface area)} \tag{1}$$

Symbols are defined in Appendix C.

Open Volume and Porosity

$$V_{open} = V_{total} - V_{solid} = P_{or}V_{total} \tag{2}$$

Geometric Parameters

(1) Number of bristles in axial and circumferential directions:

$$N_x = \frac{t_1 - d}{H_{t_1}} + 1; \qquad N_\theta = \frac{N}{N_x} \tag{3}$$

$$H_{t_1}^2 = (e_0 + d)^2 - \frac{1}{4}[e_0 + d \sec(\theta + \phi)]^2 \tag{4}$$

(2) Bristle spacing in circumferential direction:

$$e_\theta = \frac{w}{N_\theta}\left(1 + \frac{H}{2R}\right)d \sec(\theta + \phi) \tag{5}$$

(3) Transverse and longitudinal pitch (center-to-center distances between nearest neighbors in adjacent rows or columns):

$$S_T = d \sec(\theta + \phi) + e_\theta \tag{6}$$

$$S_L^2 = \left(\frac{S_T}{2}\right)^2 + H_{t_1}^2 \tag{7}$$

Leakage Parallel to Axial Path

The empirical relation for friction factor as given by Gunter and Shaw[17] is

$$\frac{f}{2} = \frac{\Delta P g D_v \rho}{10 \langle t \rangle G_1^2} \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{S_T}{D_v}\right)^{0.4} \left(\frac{S_T}{S_L}\right)^{0.6} \tag{8}$$

$$= \bar{\Phi} \frac{\Delta P \rho g}{G_1^2}$$

where

Laminar flows

$$\frac{f}{2} = \frac{90}{(Re)_v} \quad (Re)_v < 200 \quad (Re)_v = \frac{D_v G_1}{\mu}$$

Turbulent flows

$$\frac{f}{2} = \frac{0.96}{(Re)_v^{0.145}} \quad W_1 = G_1 \langle A_\perp \rangle \tag{9}$$

Leakage Normal to Axial Path

$$W_3 = (C_d)_\parallel A_\parallel \sqrt{2\rho \, \Delta P} \tag{10}$$

$$(C_d)_\parallel = S \sqrt{2\bar{\Phi}/f} \tag{11}$$

For the 130-mm (5.1-in.) diameter seal, $S = 1$ and $(C_d)_\parallel = 0.4$ for flows less than 0.75 scfm and $(C_d)_\parallel = 0$ for flows greater than 0.75 scfm. For the 38-mm (1.5-in.) diameter seal, $S = 1/5$ and $(C_d)_\parallel = 0.2$.

Leakage at Interface

Assuming bristle rows to be similar to sequential orifices, the results from Gunter and Shaw[17] can be applied and the axial flow through the clearance with N_x bristle rows becomes

$$G_2 = 0.2 A_c \sqrt{2\rho \, \Delta P} = \frac{W_2}{wC} \tag{12}$$

Seal Leakage

$$\dot{m} = W_1 + W_2 + W_3 \tag{13}$$

Interpretation of W_3 Leakage

In order to illustrate some relationships between leakage through the brush and that along the fibers, a simplified laminar model was selected as it provides linear relationships between pressure drop and mass flow. These relationships are not used in the analysis herein but will be expanded upon in a later paper.

Perpendicular to Fiber

$$\phi_\perp = \frac{64\sqrt{1 - P_{or}}}{P_{or}} \left[1 + 14.75(1 - P_{or})^3\right] \tag{14}$$

Parallel to Fiber

$$\phi_\parallel = \frac{15.74(1 - P_{or})^{1.43}}{P_{or}} \left[1 + 27(1 - P_{or})^3\right] \tag{15}$$

Mass flow and pressure drop are related to geometry as follows:

$$\phi_T = \frac{\rho \, \Delta P d^2 A}{\mu \langle t \rangle \dot{m}} \tag{16}$$

Let the flow areas and the relationship between resistances be defined as

$$A_\perp = \frac{V_{open}}{\langle t \rangle} = P_{or} H w \tag{17}$$

$$A_\parallel = \frac{V_{open}}{\langle H_f \rangle} = P_{or} \langle t \rangle w \frac{H}{\langle H_f \rangle} \tag{18}$$

$$\phi_\perp = M \phi_\parallel \quad 2 < M < 8 \tag{19}$$

These relationships are now expressed in terms of flow resistance.

6

Flow Resistance Perpendicular to Fiber

$$R_\perp = \left(\frac{\phi_\perp \mu}{\rho d^2}\right)\left(\frac{\langle t \rangle}{P_{or} H}\right) \qquad (20)$$

Flow Resistance Parallel to Fiber

$$R_\parallel = \left(\frac{\phi_\parallel \mu}{\rho\, d^2}\right)\left(\frac{\langle H_f \rangle}{P_{or} \langle t \rangle}\right) \qquad (21)$$

Flow Resistance of System

$$\frac{1}{R_f} = \left(\frac{\langle H_f \rangle}{M\langle t \rangle} + \frac{\langle t \rangle}{\langle H_f \rangle}\right) \Big/ (R_f)_0 \qquad (22)$$

$$(R_f)_0 = \frac{\phi_\parallel \mu}{\rho d^2 P_{or}} = \frac{\phi_\parallel \nu}{d^2 P_{or}} \qquad (23)$$

This simplified model shows that

(1) For a thick brush leakage along the bristles is important.

(2) Over a selected range in $\langle H_f \rangle / \langle t \rangle$ and M the leakage would remain nearly constant.

(3) Interface leakage enhances both normal and parallel leakage.

Appendix B

Videotape Scenes: Brush Seal Rotor-Stator Interface Dynamics

1. Sketch of apparatus for bristle deformation (130- and 38-mm (5.1- and 1.5-in.) diameter seals) shows force actuator on the brush seal perimeter with extensometer to monitor displacements. Still and video photographs were taken.

2. Bristle deformation of both 130- and 38-mm (5.1- and 1.5-in.) diameter seals for a given radial displacement shows that the bristles tended to slide relative to one another. The tip displacement diminished as $\cos(\beta + \phi)$, where β is the angle with respect to the line of radial displacement. So also then did bristle displacement, which in turn relates to the amount of sliding.

3. Scene of drill press apparatus with Margaret, Julie, and Jim provides an overall view of the system.

4. Map of flow versus pressure drop to mark where scenes are taken indicates the pressure and the anticipated leakage for various scenes presented on the tape.

5. 0-rpm pressure ramp (top lighting and halogen lamp effects): As the pressure increased, the stator was permitted to move relative to the rotor, resulting in a shift of the interface. A clearance opened on one side and the bristles were compacted on the other. Light patterns formed at the interface show various degrees of nonuniformity at the interface. In some cases "holes" were noted.

6. Map of flow versus pressure drop to mark where scenes are taken is repeated here. Again it indicates the pressure and the anticipated leakage for various scenes presented on the tape.

7. 400-rpm pressure ramp (top lighting and halogen lamp effects): Although it is clear that the bristles responded at 400 rpm, it is not clear if there was phase shift. Light patterns tend to indicate faithful contact at low pressures. But as the pressure increased, angular movement of the stator opened clearance on one side and compressed the bristles on the other.

8. Summary: No-flow simple mechanical loading indicated that bristle motion was principally sliding and packing with some potential for bending. The action appeared to be uniform.

(1) The brush rotor-stator interface showed a variety of nonuniform contact patterns.

(2) At low rotor speed bristles followed eccentric rotor motions.

(3) Interface nonuniformities increased with pressure drop.

(4) The optical system provided a good method for determining clearances at the interface.

Appendix C

Nomenclature

A	flow area
A_c	flow through clearance
$A_\perp$	flow area normal to bristles
$A_\parallel$	flow area parallel to bristles
C	clearance
$(C_d)_\parallel$	flow coefficient parallel to bristles
D_v	volumetric hydraulic diameter
d	bristle diameter
e_θ	bristle spacing circumferential direction
e_0	bristle manufacturing spacing
f	friction factor
G_1	mass flux through bristles
g	units conversion constant
H	seal dam height (fence height)
$\langle H_f \rangle$	effective seal dam height (fence height)
L	bristle length
M	constant
$\dot{m}$	mass flow
N	number of bristles
N_x	number of bristles in axial direction
N_θ	number of bristles in circumferential direction
P_{or}	porosity based on geometry
ΔP	pressure drop
R	shaft radius
R_f	flow resistance
$(R_f)_0$	flow resistance parameter
$R_\perp$	flow resistance (flow normal to bristles)
$R_\parallel$	flow resistance (flow parallel to bristles)
$(Re)_v$	volumetric Reynolds number
S	flow coefficient parameter
S_L	lateral pitch
S_T	transverse pitch
t_1	compact brush thickness
t_2	brush thicknesses at H
t_3	brush thicknesses at interface
$\langle t \rangle$	average brush thickness
V_{open}	open volume
V_{solid}	solid volume
V_{total}	total volume
W_1	mass flow rate through bristles
W_2	mass flow rate through clearance

W_3	mass flow rate along bristles near fence
w	seal width, $R\phi_0$
ρ	density
θ	bristle angle
μ	viscosity
μ_w	viscosity at interface
ν	μ/ρ
Φ	friction parameter
ϕ	shaft angle to bristle
$\phi_\perp$	geometry factor (flow normal to bristles)
$\phi_\parallel$	geometry factor (flow parallel to bristles)
ϕ_T	geometry factor (overall)
ϕ_0	reference angle for friction surface, w/R
ω	rotational speed

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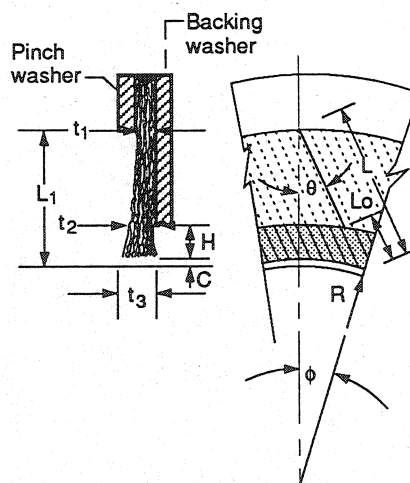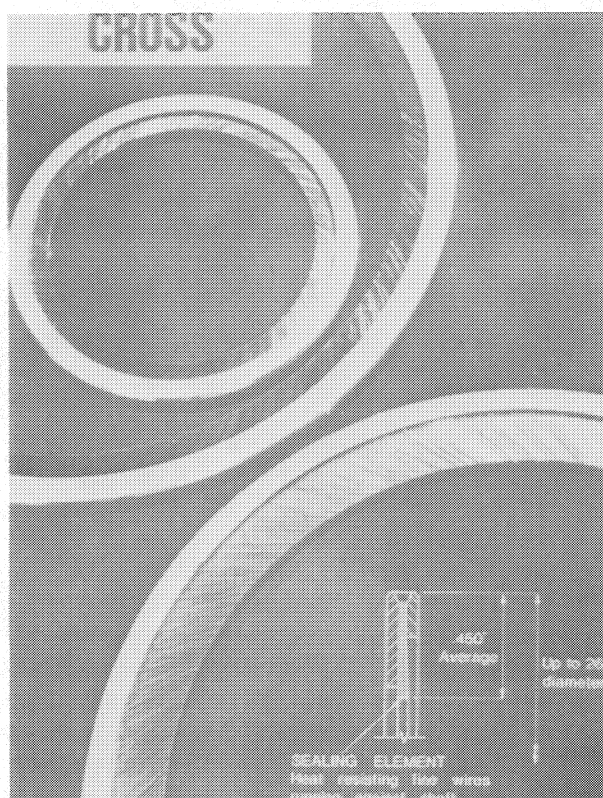
Figure 1.—Geometry of circular brush seal. (Courtesy of Cross Mfg. Ltd.)
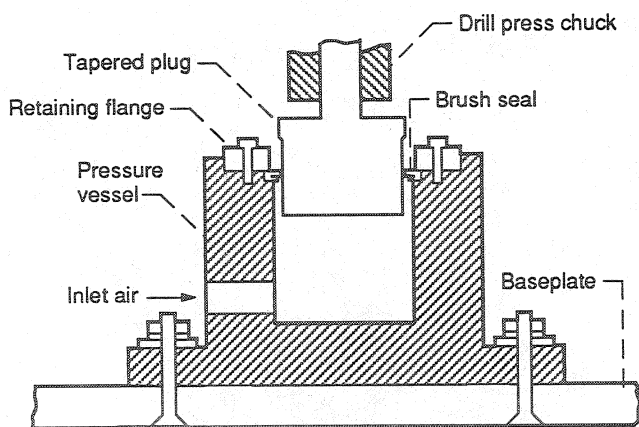
Figure 2.—Test apparatus.
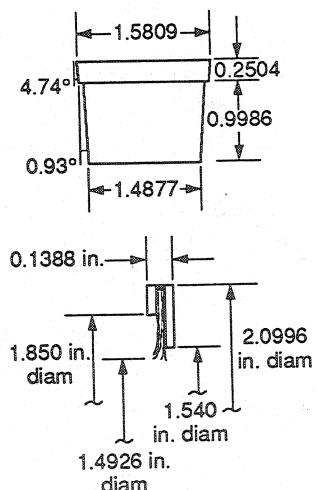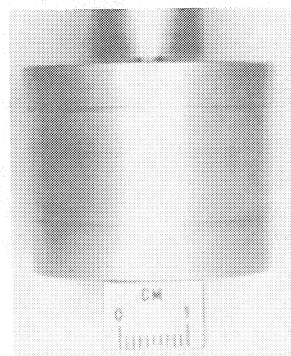
Figure 4.—Tapered plug.
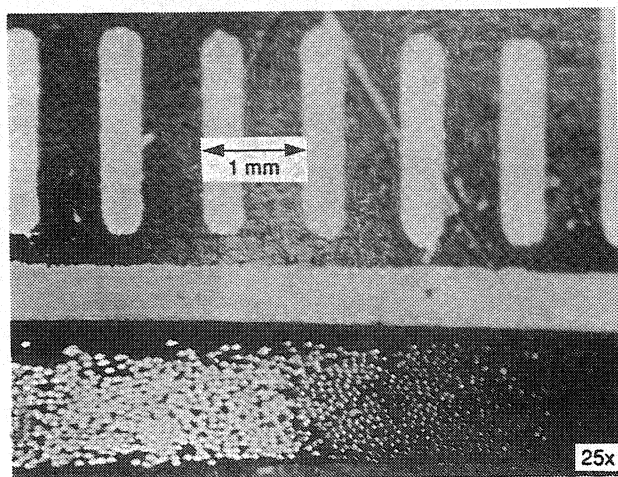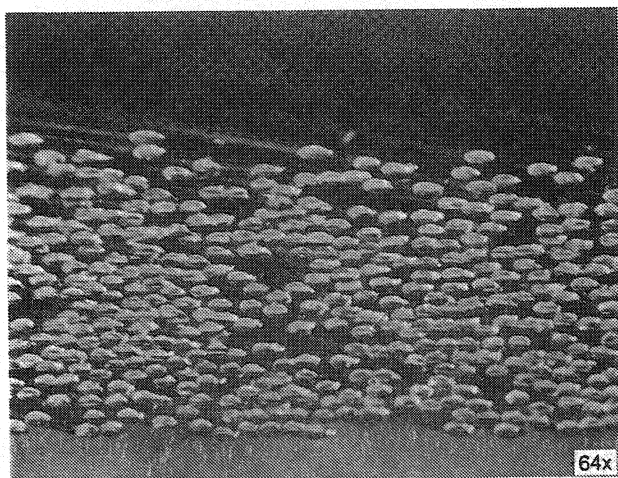
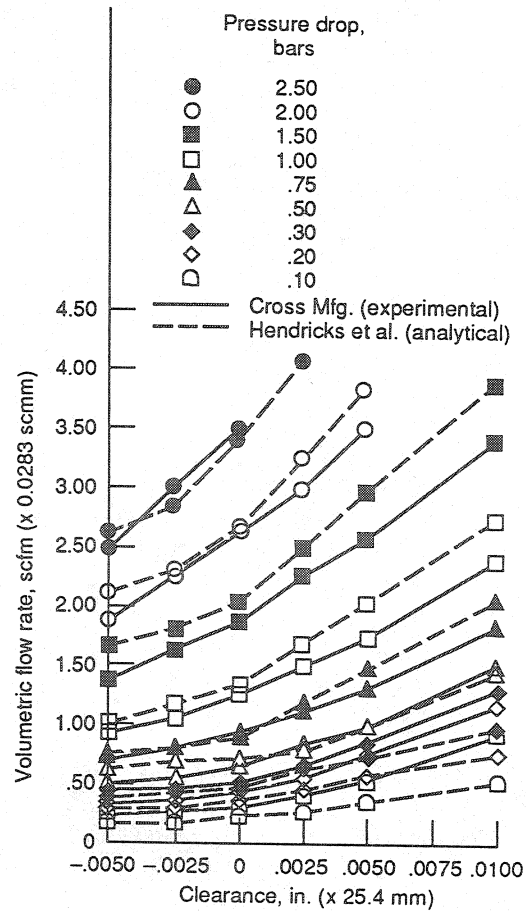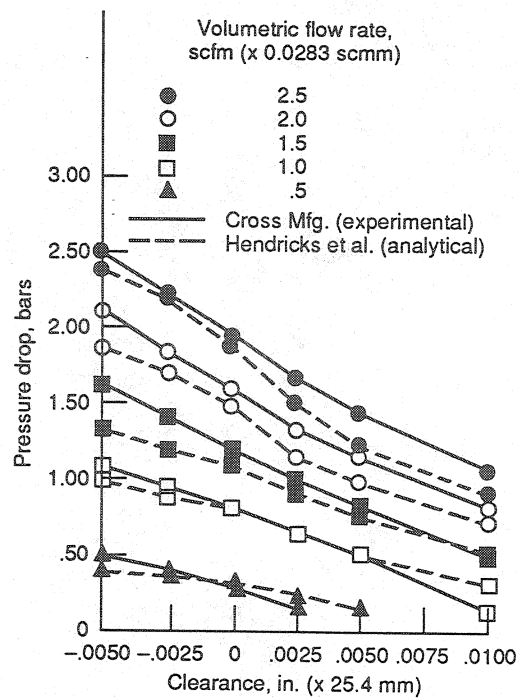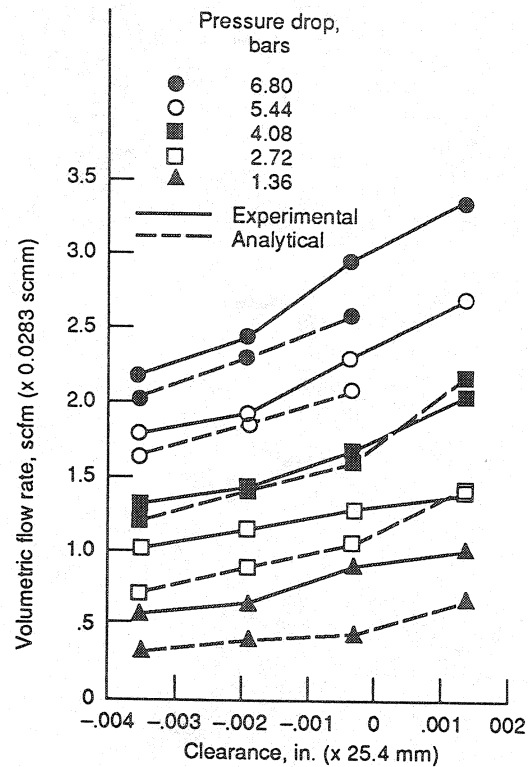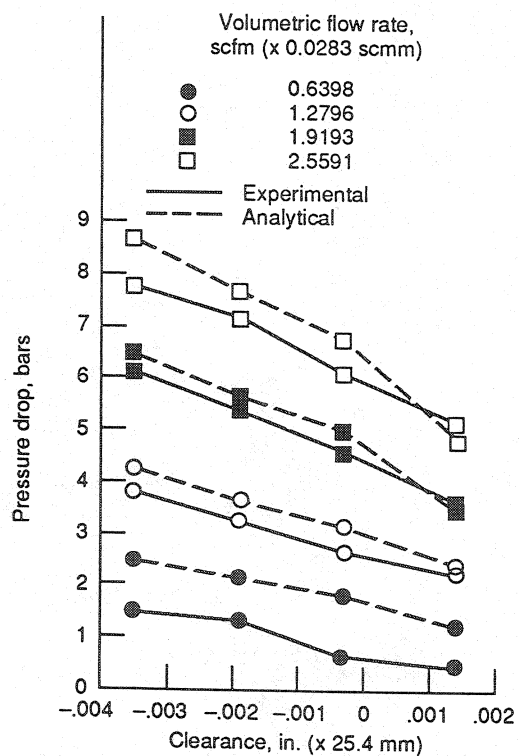Figure 3.—Circular brush seal.
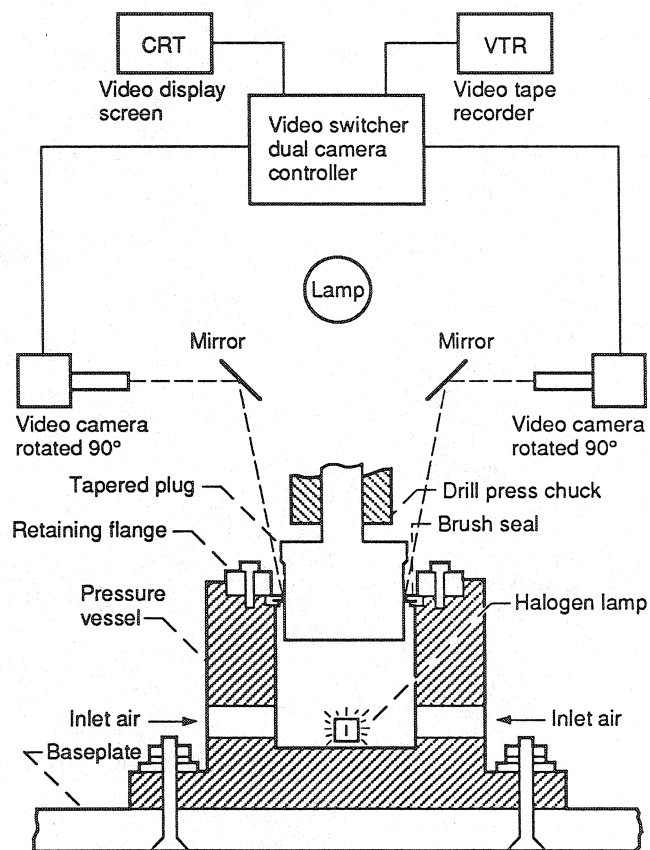
Figure 5.—Test setup.

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(a) 130-mm (5.1- in.) diameter brush seal.

(b) 38-mm (1.5- in.) diameter brush seal.

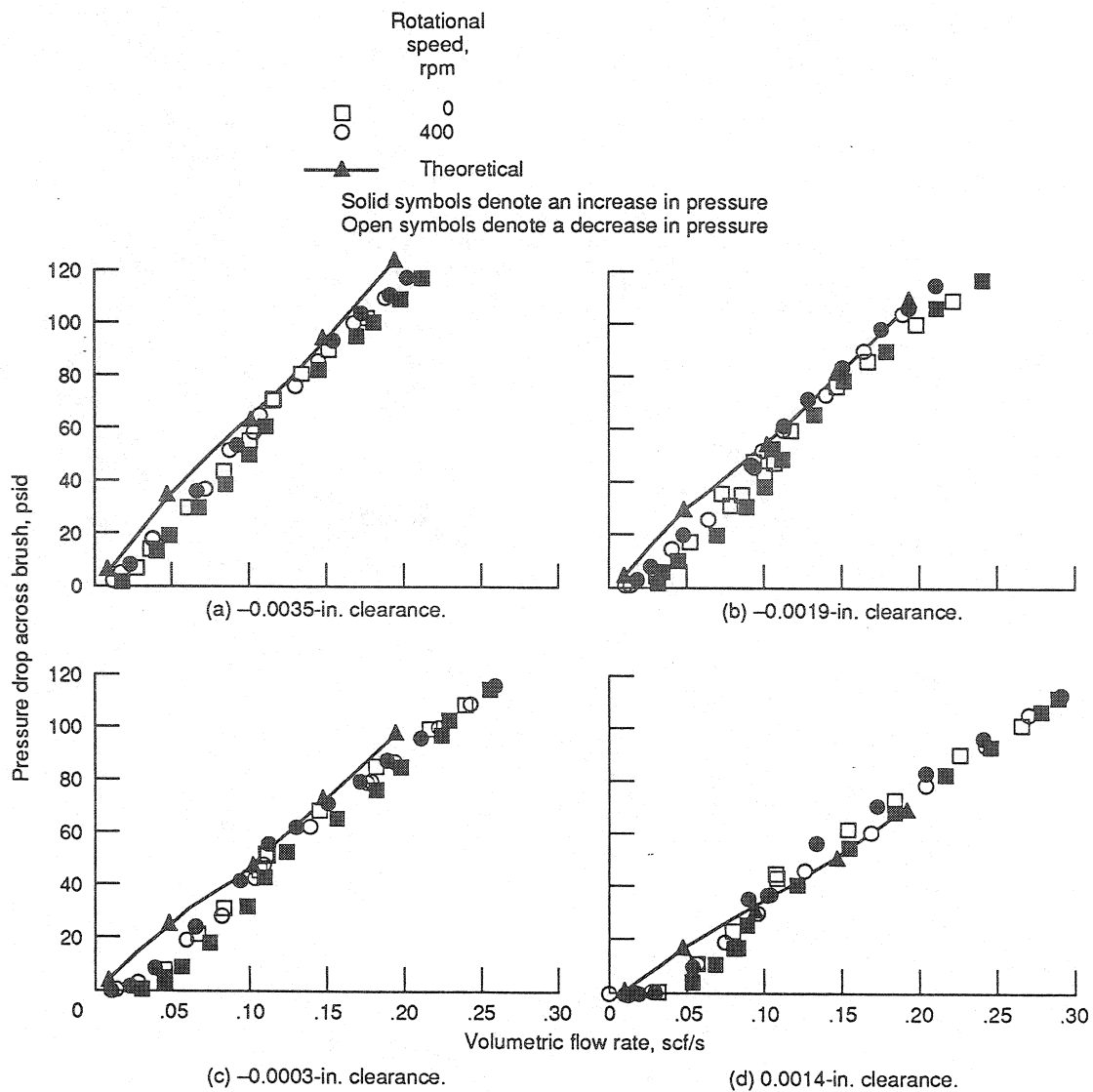Figure 6.—Analytical predictions of flow and pressure drop as functions of clearance.

Figure 7.—Comparison of data at 0 and 400 rpm without flow straightener.
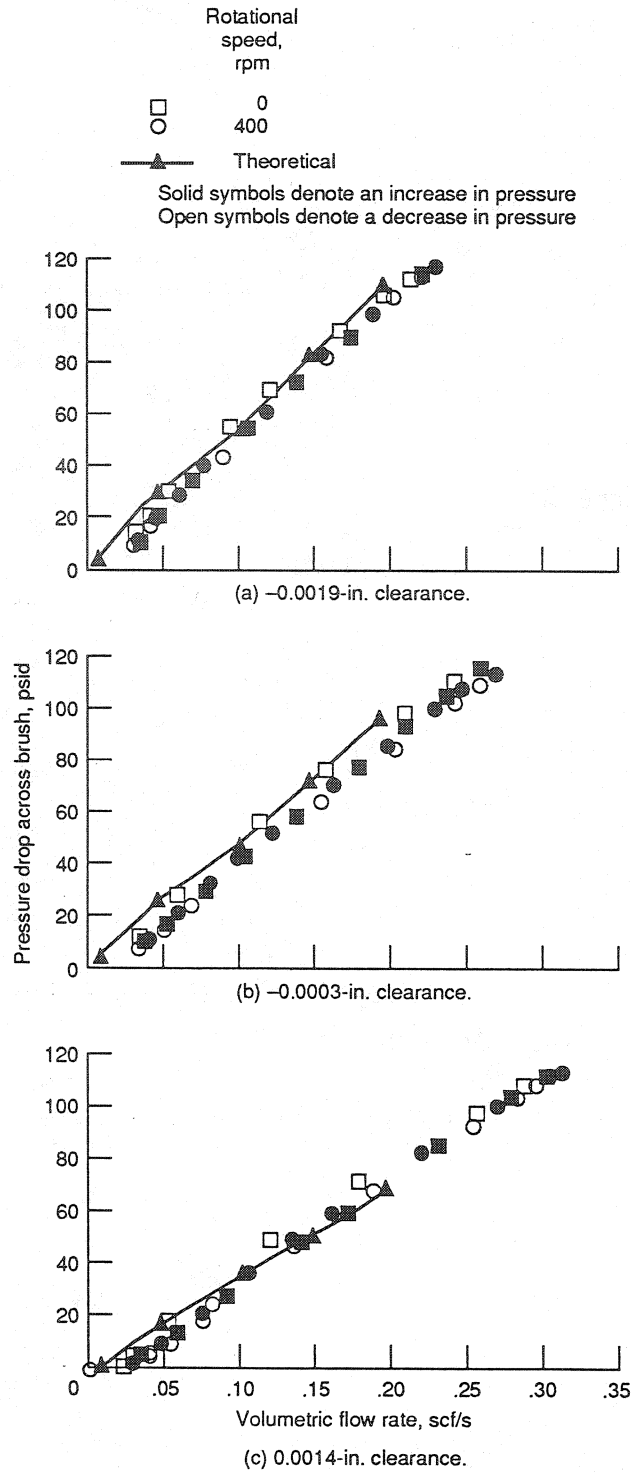
Figure 8.—Comparison of data at 0 and 400 rpm with flow straightener.

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13. ABSTRACT (Maximum 200 words)

Some preliminary brush seal leakage results for ambient-temperature air are presented. Data for four nominal brush-rotor radial clearances of –0.09, –0.048, –0.008, and 0.035 mm (–0.0035, –0.0019, –0.0003, and 0.0014 in.) were taken by using a tapered plug rotor at 0 and 400 rpm with rotor runout of 0.127 mm (5 mils) peak to peak. The brush seal nominal bore diameter was 38 mm (1.5 in.) with 0.05-mm (2-mil) bristles at 200 bristles/mm of circumference (5000 bristles/in. of circumference) and a 0.61-mm (24-mil) fence height. Leakages were greater than predicted, but agreement was reasonable. Leakage rates were not significantly altered by hysteresis or inlet flow variations. Visualization studies showed that the bristles followed the 400-rpm excitation, and loading studies indicated that bristles slid relative to one another.

14. SUBJECT TERMS

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